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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

No. 427

STRENGTH TESTS ON THIN-WALLED DURALUMIN CYLINDERS IN TORSION

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SUMMARY

This report is the first of a series presenting the results of strength tests on thin-walled cylinders and truncated cones of circular and elliptical section; it comprises the results obtained to date from torsion (pure shear) tests on 65 thin-walled duralumin cylinders of circular section with ends clamped to rigid bulkheads. The effect of variations in the length/radius and radius/thickness ratios on the type of failure is indicated, and a semi-empirical equation for the shearing stress at maximum load is given.

INTRODUCTION

A survey made about a year ago of the available information and the methods employed in the design of stressed-skin structures for aircraft revealed that the design of structures of this type is based largely upon the results of experience and data obtained either from tests on structures similar to the one under consideration or from tests on the particular structure under consideration. A study of the existing reports of static tests upon monocoque fuselages and stressed-skin wings resulted in the conclusion that very little reliable information of a fundamental character concerning allowable stresses could be obtained from full-scale tests already made because no mention was made of the properties of the material and because failure of the structure as a whole was caused by a variety of different types of failure in the skin, reinforcement, and connections. Consequently, the National Advisory Committee for Aeronautics, in cooperation with the Army Air Corps, Navy Bureau of Aeronautics, and the Bureau of Standards, Department of Commerce, has outlined a research program to investigate stressed-skin structures for aircraft.

As a part of this research program an extensive series of tests on thin-walled duralumin cylinders and truncated cones of circular and elliptical section is being made at Langley Field, Va. In these tests, the absolute and relative dimensions of the specimens are being varied to study the types of failure and to establish useful quantitative data in the following loading conditions: torsion, compression, bending, and combined loading.

This report is the first of a series presenting the results of these tests; it comprises the data obtained in the torsion (pure shear) tests on thin-walled duralumin cylinders. Although the tests have not as yet been completed, it was thought advisable to publish the results thus far obtained.

MATERIAL

The duralumin (Al. Co. of Am. 17ST) used in these tests was obtained from the manufacturer in sheet form in thicknesses ranging from 0.0105 to 0.0228 inch. The properties of the material as determined by the Bureau of Standards from specimens selected at random are given in Table I. Typical stress-strain curves taken longitudinally and transversely to the direction of rolling are given in Figures 1 and 2.

Upon reference to the above-mentioned table and figures it will be observed that the modulus of elasticity is substantially the same in the two directions of the sheet but that the ultimate strength and yield point are considerably lower transversely to the direction of rolling than longitudinally. However, as all the cylinders tested failed at stresses considerably below the yield-point stress, the difference in the strength properties in the two directions has no bearing on the results.

SPECIMENS

The test specimens consisted of right circular cylinders of 7.5 and 15.0 inch radius with lengths ranging from 2.25 to 45.0 inches, which were constructed in the following manner. First, a duralumin sheet was cut to the dimensions of the developed surface. The sheet was then wrapped about and clamped to the end bulkheads. (See figs.

3, 4, and 6.) With the cylinder thus assembled, a butt strap 1 inch wide and of the same thickness as the sheet was fitted, drilled, and bolted in place to close the seam. In the assembly of the specimen, care was taken to avoid having either "soft spots" or wrinkles in the walls when finally constructed.

The end bulkheads, to which the loads were applied, were each constructed of two steel plates one-fourth inch thick separated by $1\frac{1}{2}$ inches of plywood core. These parts were bolted together and turned to the specified outside diameter. Steel bands approximately one-fourth inch thick were used to clamp the duralumin sheet to the bulkheads. These bands were bored to the same diameter as the bulkheads.

APPARATUS AND METHOD

The thickness of each sheet was measured to an estimated precision of ± 0.0003 inch at a large number of stations by means of a dial gauge mounted in a special jig. The average thicknesses of the sheets were used in all calculations of radius/thickness ratio and stress.

A photograph of the loading apparatus used in the torsion tests is shown in Figure 3. The load from the jack was applied through a thrust bearing to a yoke. By means of a flexible cable that passed over a series of pulleys, half of the load was transmitted to one end of the lower horizontal beam as an up load and the other half was transmitted through the rectangular frame to the other end of the same beam as a down load. In this manner torque, unaccompanied by transverse shear, was applied to the specimen. A shear load on the cylinder caused by the weight of the apparatus, including the forward bulkhead and band, was balanced by a counterweight as shown in Figure 3.

In order to determine the possible errors caused by friction in the joints of the frame, a special test was made in which the moment applied to the lower beam was measured directly and compared with the moment calculated from the force applied by the jack. These two moments were found to agree within 1 per cent throughout the range of moments applied.

Loads were applied by the jack in increments of about 1 per cent of the estimated load at failure, except after the first few tests had been made when the first increment of load was about half the estimated load at first wrinkle. At first wrinkling one or more diagonal wrinkles began to form and grew steadily in size and number until failure occurred by a sudden increase in deformation and the formation of wrinkles in the complete circumference. (See fig. 4.) In all the tests 5 to 10 minutes elapsed from the time that load was first applied to the specimen until failure occurred.

For the very short specimens (length/radius ratios of 0.5 or less) deformation following failure was accompanied by an increase in load, whereas for all other specimens deformation following failure was accompanied by a decrease in load. (See fig. 5.) In order to insure that the recorded loads at failure for the short specimens were correct, two dial gauges were used to measure the twist of the cylinder between bulkheads (fig. 6) and the load at which a sudden increase in deformation occurred, as determined from a load deformation curve, was recorded as the load at failure.

DISCUSSION OF RESULTS

The strength of thin-walled circular tubes subjected to torsion has been treated theoretically in references 1 and 2. In reference 1 equations are given from which it is possible to calculate the critical stress but the computations are tedious and involved and have been made only for radius/thickness ratios far below those usually encountered in stressed-skin structures for aircraft. In reference 2 relatively simple equations are given for the critical stress but these could not be checked by the results herein reported. Consequently, no consideration is given to the correlation of the experimental results with theory in this report. However, when the general investigation of the strength of thin-walled cylinders is completed this correlation will be attempted.

The results shown in Figures 7, 8, and 9 are self-explanatory. They show that for geometrically similar cylinders, the number of shear wrinkles, the angle between the wrinkles and the cylinder elements when the wrinkles start to form, and the shearing stress at failure, are constant. For any given value of the radius/thickness ratio, the number of shear wrinkles, the angle between the wrinkles and the cylinder elements when the wrinkles start to form, and the shearing stress at failure all decrease with increase in the length/radius ratio. For any given value of the length/radius ratio, the number of shear wrinkles increases with increase in the radius/thickness ratio, but the angle between the wrinkles and the cylinder elements when the wrinkles start to form and the shearing stress at failure both decrease.

As first wrinkling occurred in many of the specimens before failure, shearing stress at first wrinkle as a percentage of shearing stress at failure has been plotted against the length/radius ratio in Figure 10. It will be noted that the points scatter widely but show a general tendency for first wrinkle to occur at a lower percentage of stress at failure for the thinner sheets than for the thicker sheets. This fact is attributed to the presence of waves and dents in the thinner sheets which caused the cylinders constructed from this material to be, relatively speaking, less perfect than cylinders constructed from the thicker material.

As the experimental points for shearing stress at failure in Figure 9 plot along smooth curves irrespective of the wide scattering of the points for stress at first wrinkle in Figure 10, it is concluded that the presence of first wrinkling did not reduce the strength at failure to any appreciable extent.

An examination of the data plotted in Figure 9 indicated that the shearing stress at failure could be given very closely by an equation of the general form

$$S_s = \frac{K}{\left(\frac{r}{t}\right)^n} \quad (1)$$

where, for a given material, n is a constant and K is a function of the length/radius ratio of the cylinder. In this equation K has the dimensions of a stress. As theoretical calculations always show that the stress at which elastic instability occurs varies directly with E, the modulus of elasticity, equation (1) will be written in the following form:

$$S_s \doteq \frac{K_s\ E}{\left(\frac{r}{t}\right)^n} \quad (2)$$

where K_s is a nondimensional constant. In all the calculations for the evaluation of K_s and n, the secant modulus for S_s, the stress at failure, is substituted for E.

The numerical value of n is established by the negative slope of straight lines on the logarithmic plot of S_s/E against r/t in Figure 11. Actually, the value of n is not constant but varies somewhat with length/radius ratio. For the purpose of this report, however, an average value of 1.35 was used.

Values of K_s for $n = 1.35$ have been calculated for each test and plotted in Figure 12. In this figure the greatest dispersion of the points occurs at a length/radius ratio of 1.0 for the 7.5-inch-radius cylinders with radius/thickness ratios between 630 and 700. Consequently, the dispersion of all other points may be considered to be within the experimental error.

Strictly speaking, equation (2) with the values of n and K_s derived from these tests applies only to duralumin cylinders for the range of lengths, radii, and thicknesses tested. It is now planned to extend the tests upon duralumin cylinders and also to make a few tests upon steel cylinders in order to generalize the conclusions or to make such modifications to them as may be necessary, particularly at low values of the length/radius ratio.

CONCLUSIONS

1. For geometrically similar cylinders, the number of shear wrinkles, the angle between the wrinkles and the cylinder elements when the wrinkles start to form, and the shearing stress at failure, are constant.

2. For any given value of the radius/thickness ratio, the number of shear wrinkles, the angle between the wrinkles and the cylinder elements when the wrinkles start to form, and the shearing stress at failure all decrease with increase in the length/radius ratio.

3. For any given value of the length/radius ratio, the number of shear wrinkles increases with increase in the radius/thickness ratio, but the angle between the wrinkles

and the cylinder elements when the wrinkles start to form and the shearing stress at failure both decrease.

4. The presence of slight imperfections in the cylinder may cause wrinkling to occur at loads considerably below the load at failure, but these imperfections apparently do not reduce the strength at failure to any appreciable extent.

5. The shearing stress at failure for thin-walled duralumin cylinders may be given very closely by an equation of the form

$$S_s = \frac{K_s\ E}{\left(\frac{r}{t}\right)^n}$$

where n is a constant and K_s varies with the length/radius ratio, l/r. As a tentative value, n may be assumed equal to 1.35. Values of K_s corresponding to this value of n are given by the faired curve in Figure 12 or by the following table:

l/r	0.2	0.25	0.3	0.4	0.5	0.75
	1.0	1.5	2.0	3.0	4.0	5.0
K_s	3.3	2.75	2.45	2.02	1.78	1.45
	1.27	1.06	.94	.78	.68	.61

Langley Memorial Aeronautical Laboratory,
National Advisory Committee for Aeronautics,
Langley Field, Va., July 25, 1932.

REFERENCES

1. Schwerin, E.: Die Torsions - Stabilität des Dünnwandigen Rohes. Z.f.a.M.M., Vol. V, No. 3, pp. 235-243, incl. June, 1925.

2. Sezawa, Katsutada, and Kubo, Kei: The Buckling of a Cylindrical Shell under Torsion. Report No. 76, Aero. Research Inst., Tokyo Imperial Univ., Dec., 1931.

TABLE I. PROPERTIES OF SHEET DURALUMIN USED IN STRENGTH TESTS ON THIN-WALLED CYLINDERS

Material		Ultimate tensile strength (lb. per sq.in.)		Tensile yield point (lb. per sq.in.)		Elongation in 2 inches (per cent)		Modulus of Elasticity			
								Secant modulus at a stress of 5,000 lb. per sq.in. (lb. per sq.in.)		Secant modulus at a stress of 20,000 lb. per sq.in. (lb. per sq.in.)	
Lot	Specimen No.	Longitudinal	Transverse	Longitudinal	Transverse	Longitudinal	Transverse	Longitudinal	Transverse	Longitudinal	Transverse
1	17	58,500	55,500	41,400	36,500	18	16.5	10,620,000	10,720,000	10,510,000	10,520,000
1	34	59,400	56,600	42,500	37,100	15	16.5	10,720,000	10,770,000	10,580,000	10,570,000
2	48		57,300		36,800		16.5		10,570,000		10,340,000
2	50		56,800		35,800		13.0		10,550,000		10,380,000
2	55		58,200		36,500		18.0		10,470,000		10,270,000
2	70		57,800		35,800		16.0		10,560,000		10,440,000
3	2A		62,850		39,750		19.3		10,270,000		10,130,000
3	14A		62,800		41,000		14.0		10,410,000		10,190,000
3	46A		61,550		38,750		19.0		10,550,000		10,350,000
3	63A		61,400		39,050		18.5		10,110,000		10,060,000
3	68A		61,400		37,900		18.5		10,410,000		10,070,000

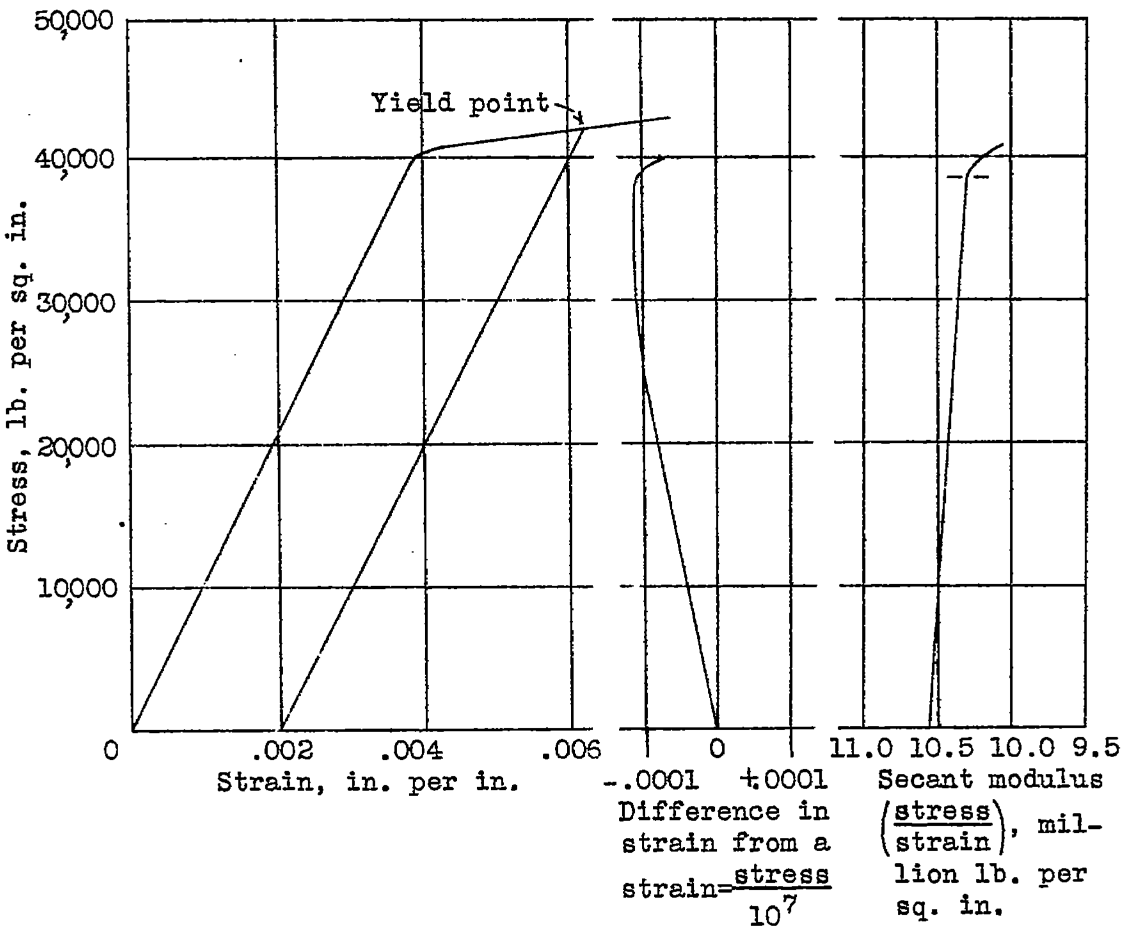


Fig. 1 Stress - strain curve for sheet duralumin 0.011 inch thick parallel to the direction of rolling.

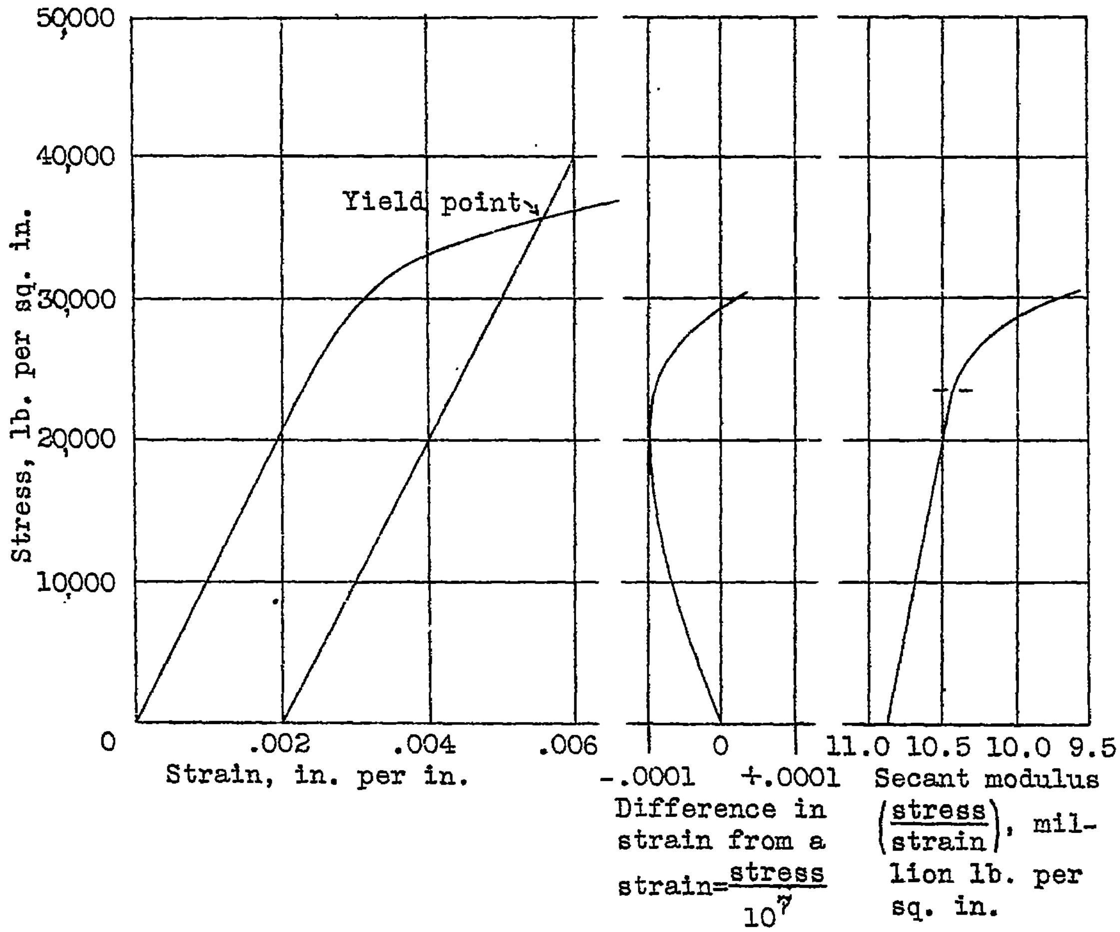


Fig.2 Stress - strain curve for sheet duralumin 0.011 inch thick normal to direction of rolling.

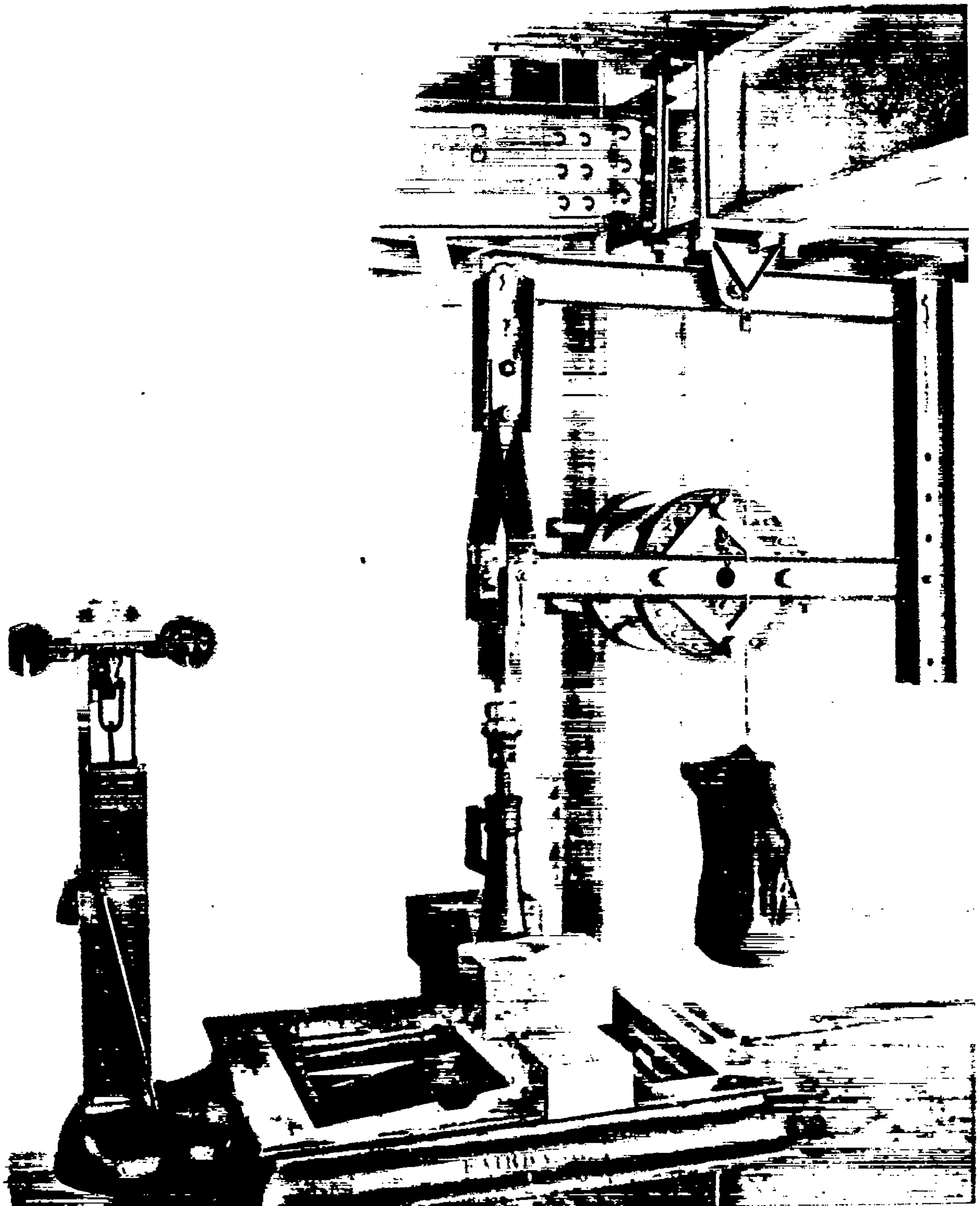
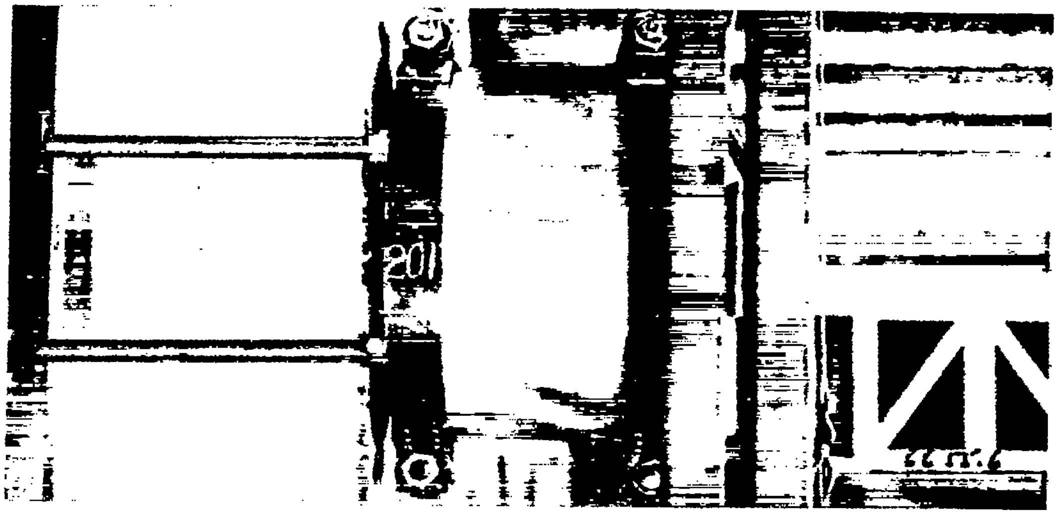


Fig. 3 Loading apparatus used in torsion tests.



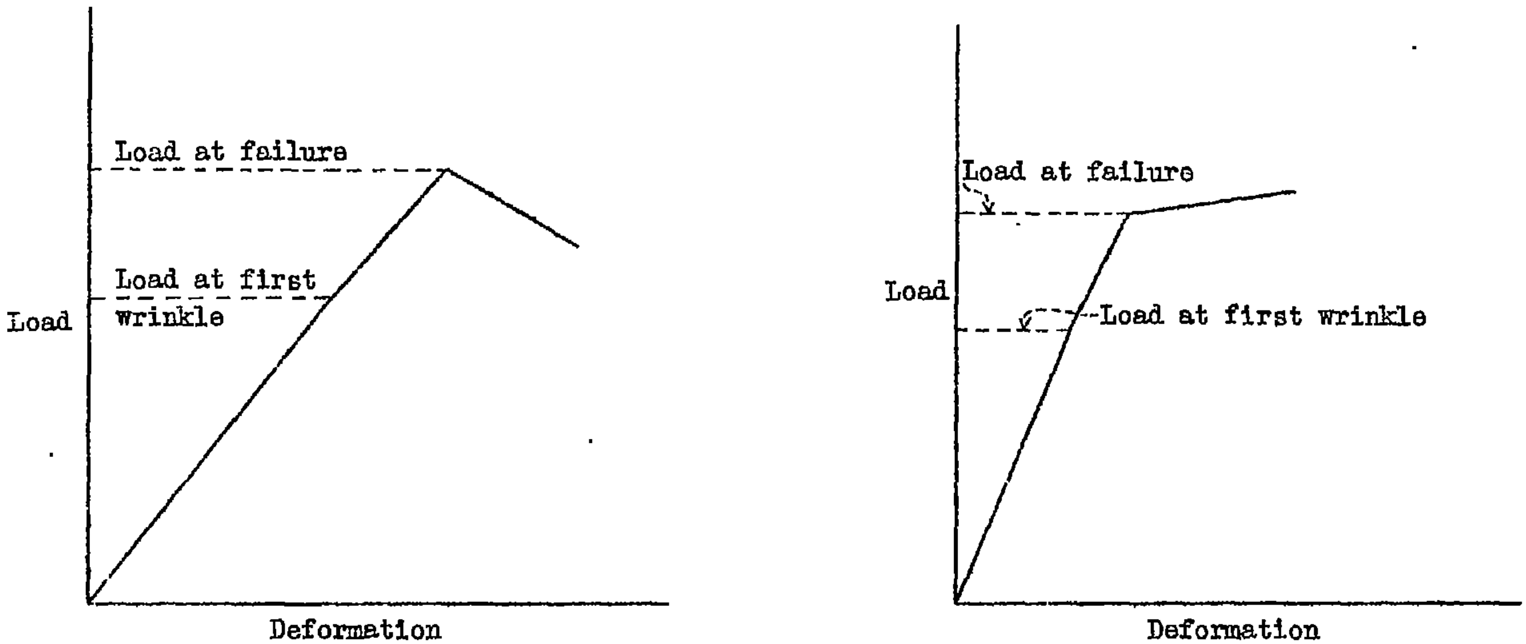
$\frac{l}{r} = 0.87$

$\frac{l}{r} = 2.0$

Fig. 4

Photographs of cylinders after failure.

$\frac{l}{r} = 3.0$



Long cylinders   Short cylinders

Fig.5 Load-deformation curves for long and short cylinders.

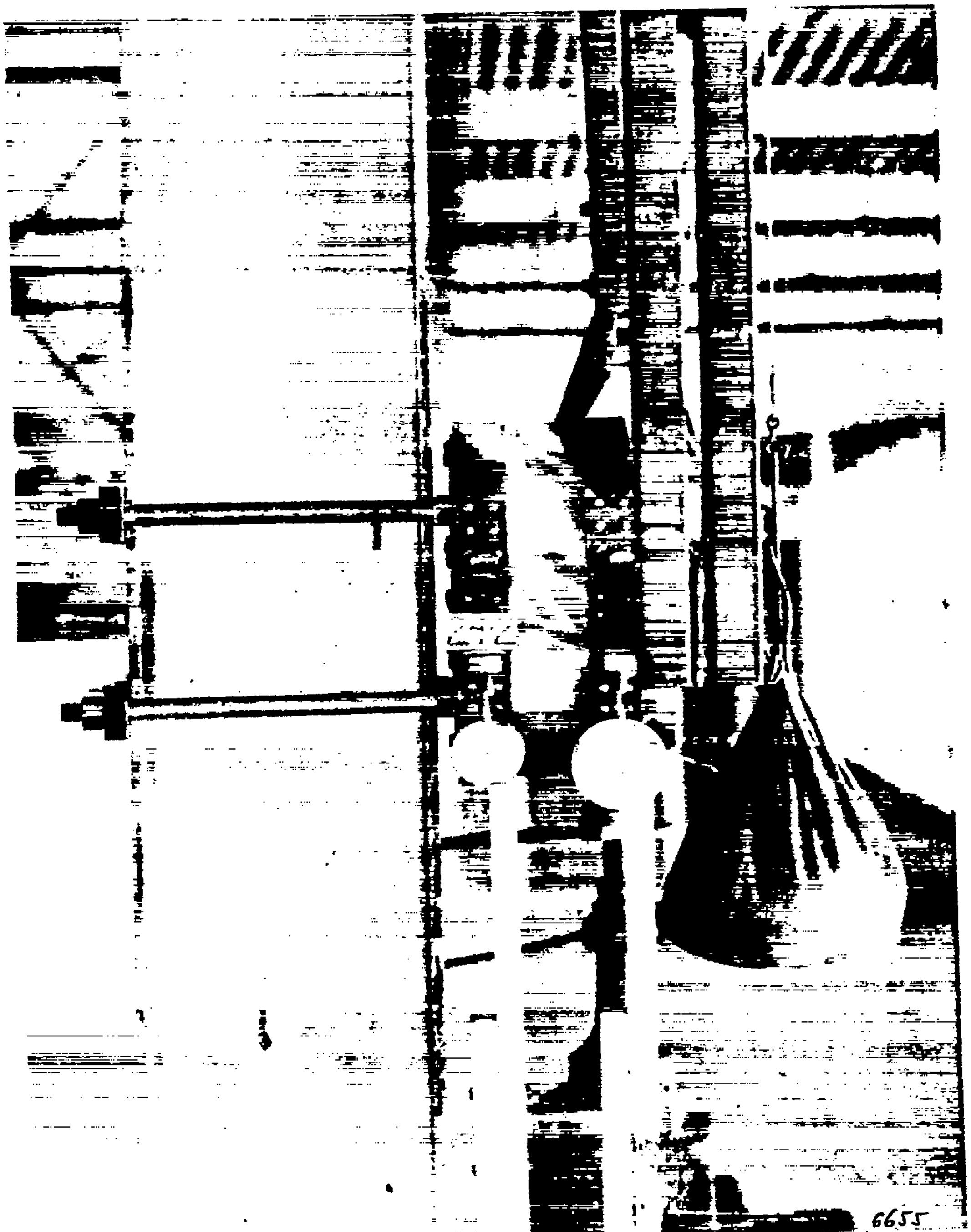


Fig. 6 Apparatus showing dial gauges used to measure the twist of the cylinder between bulkheads.

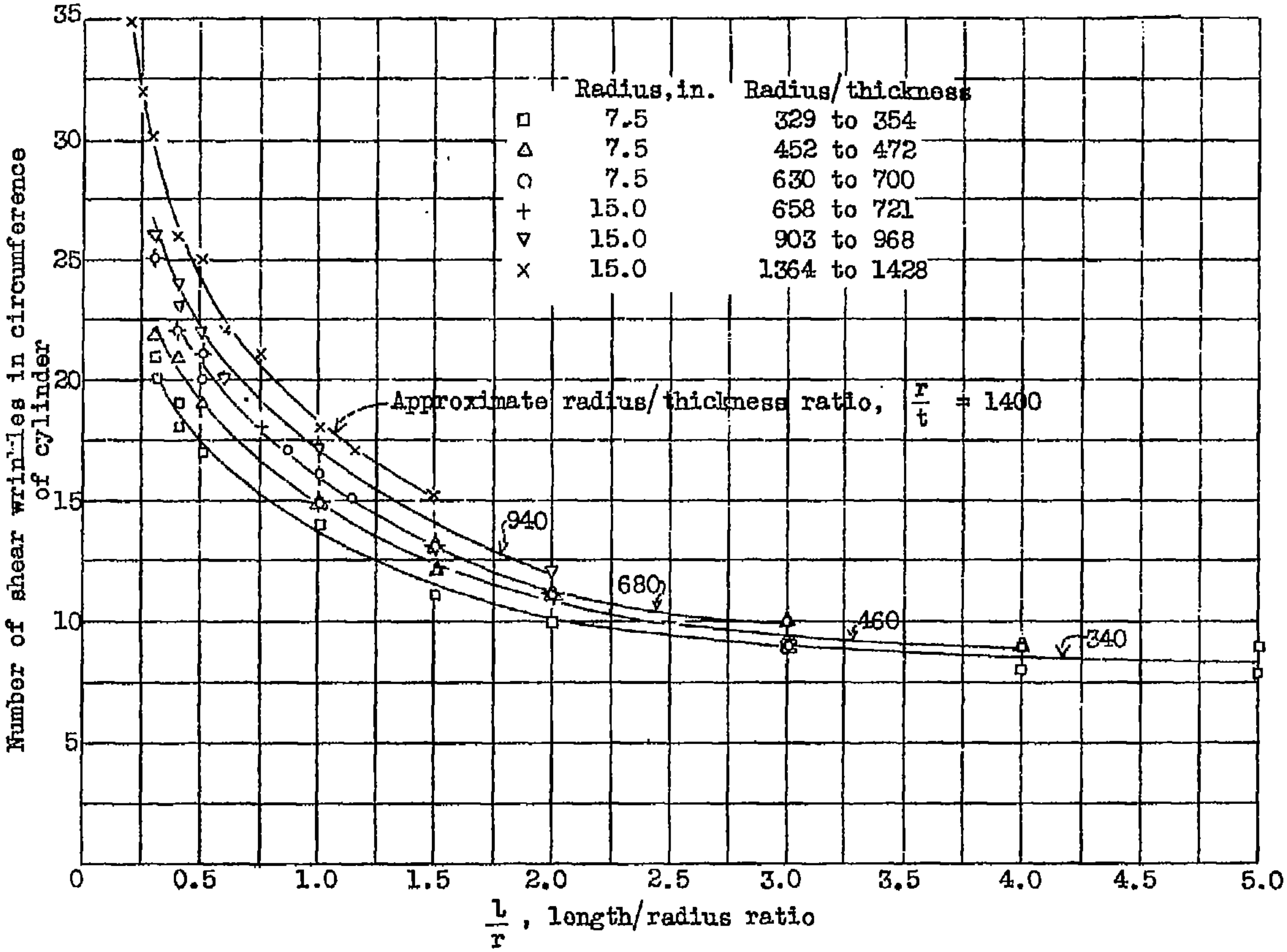
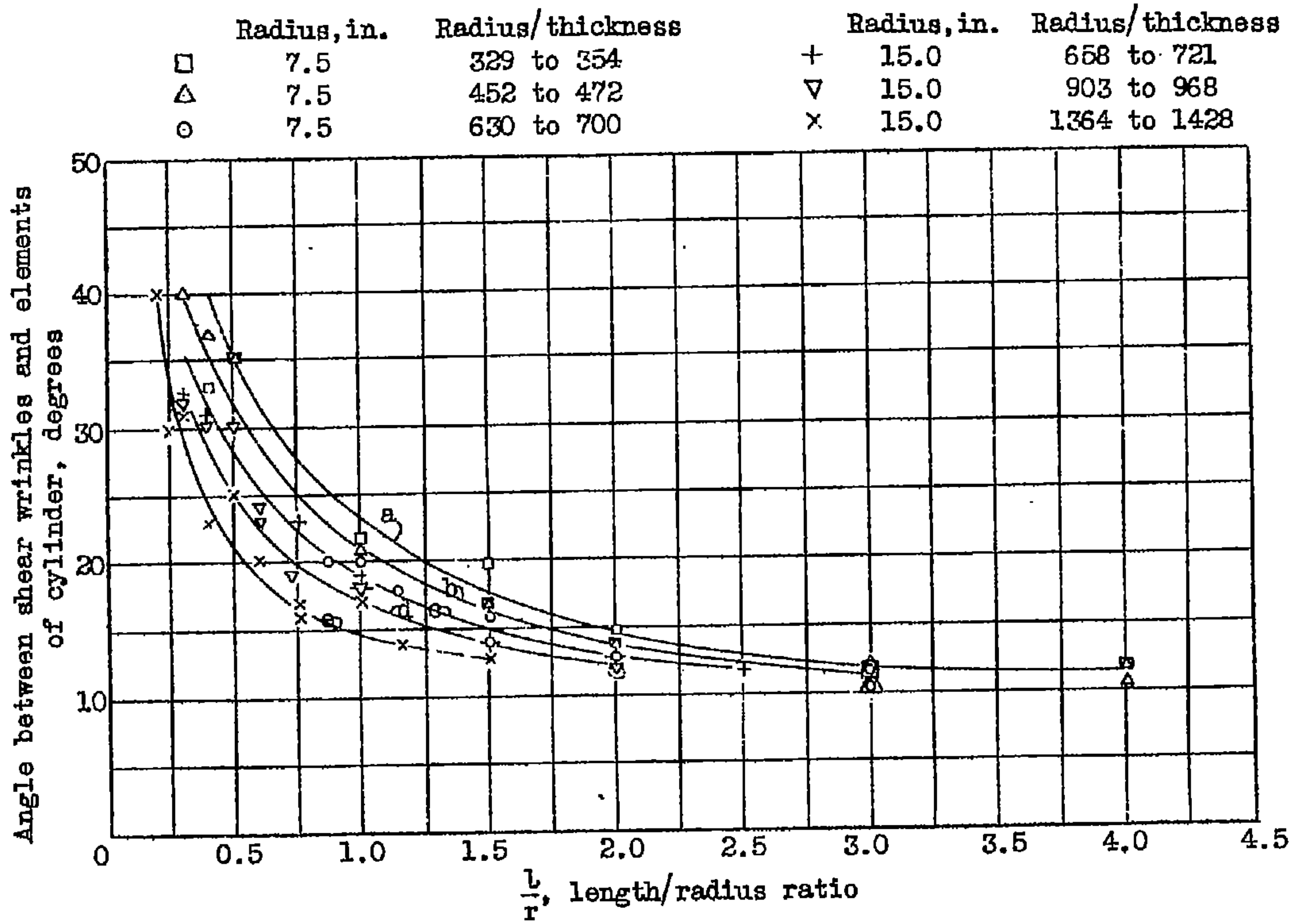
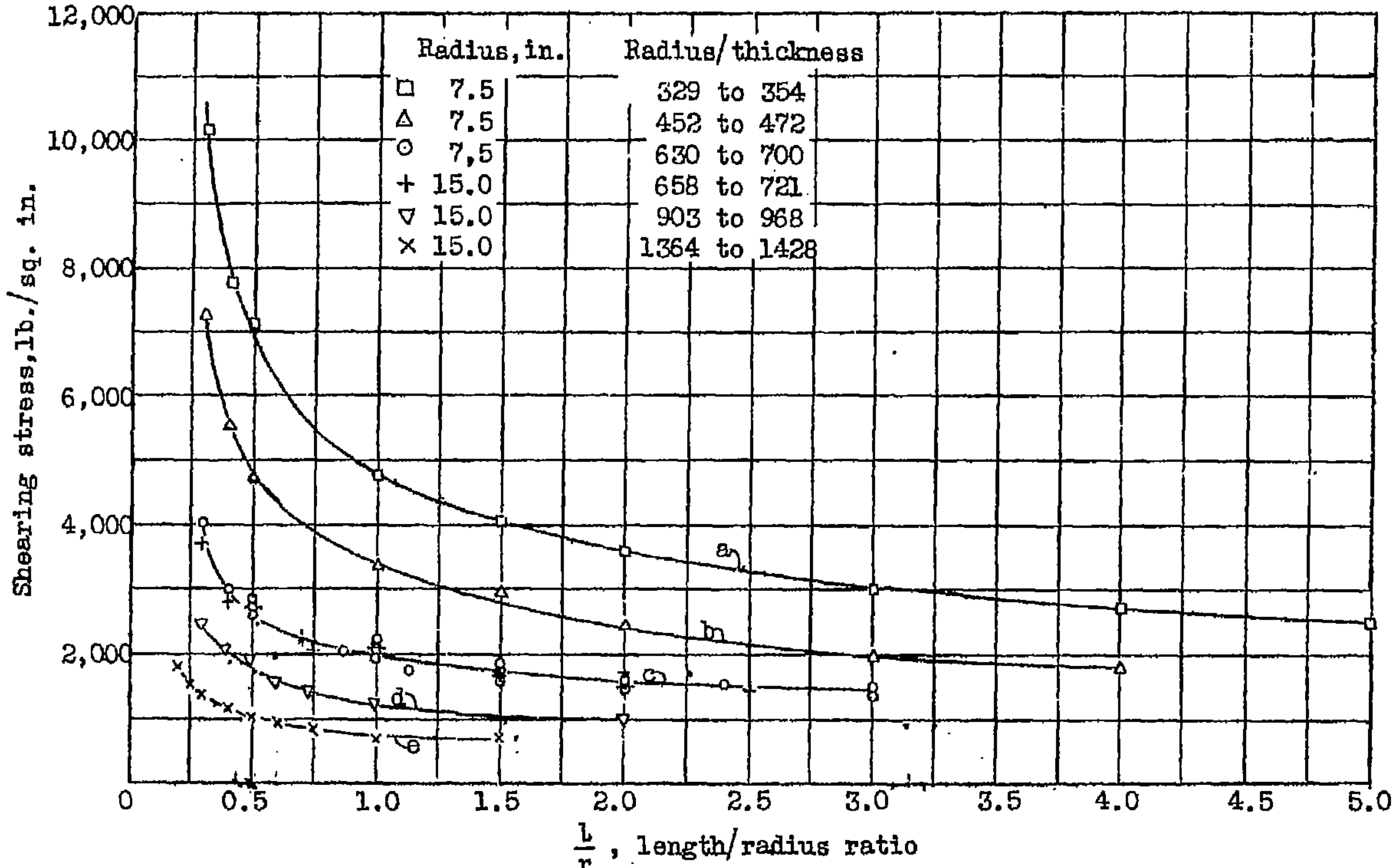


Fig.7 Number of shear wrinkles in circumference of cylinder.



a, Approximate radius/thickness ratio, $\frac{r}{t}$ =340
b, " " " " , " =460
c, " " " " , " =680
d, Approx. radius/thickness ratio, $\frac{r}{t}$ =940
e, " " " " , " =1400

Fig.8 Angle between shear wrinkles and elements of cylinder.



a, Approximate radius/thickness ratio, $\frac{r}{t}$ = 340
b, " " " " , " = 460
c, " " " " , " = 680
d, " " " " , " = 940
e, " " " " , " = 1400

Fig.9 Shearing stress at failure.

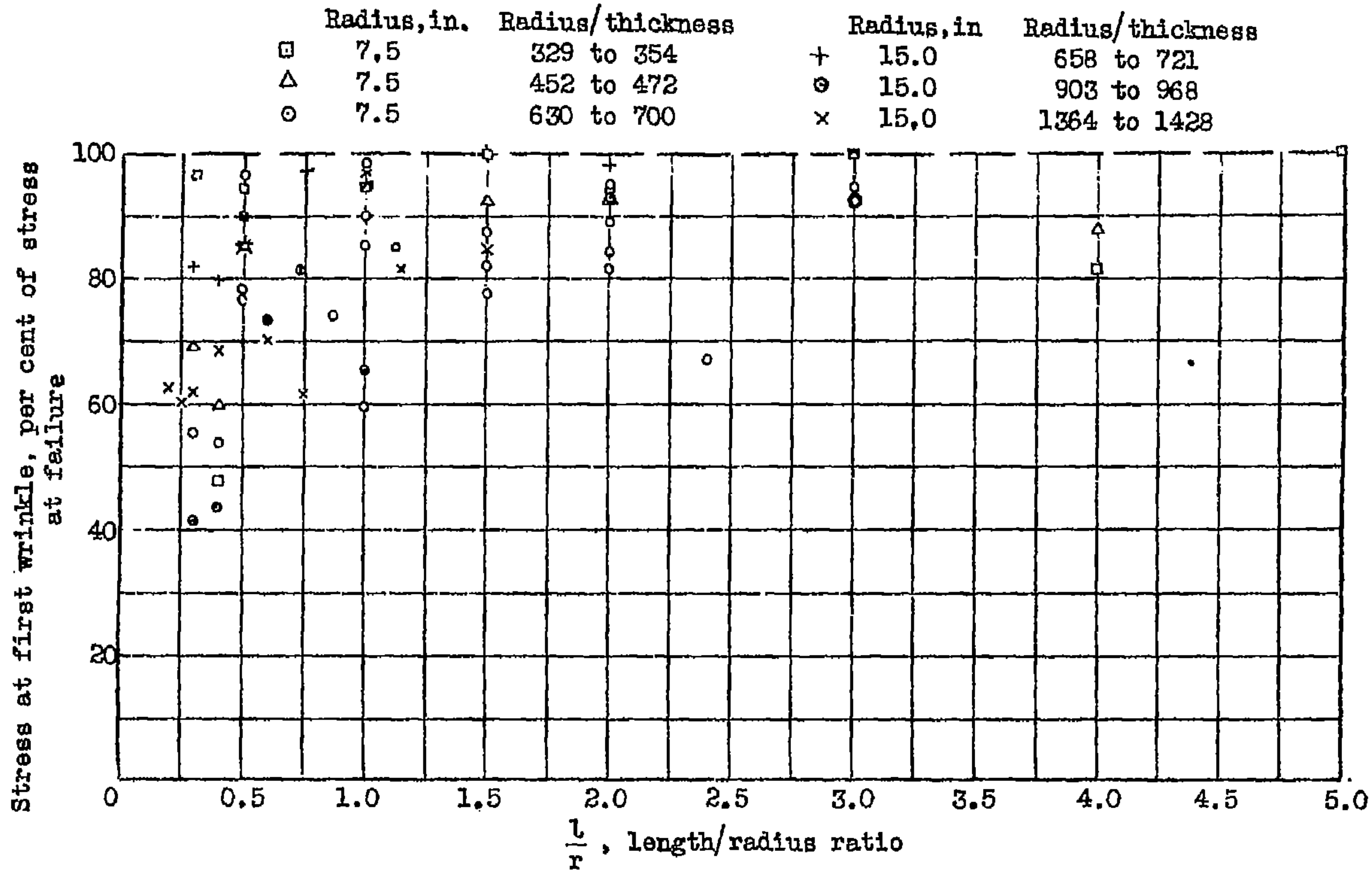


Fig.10 Shearing stress at first wrinkle as a percentage of shearing stress at failure.

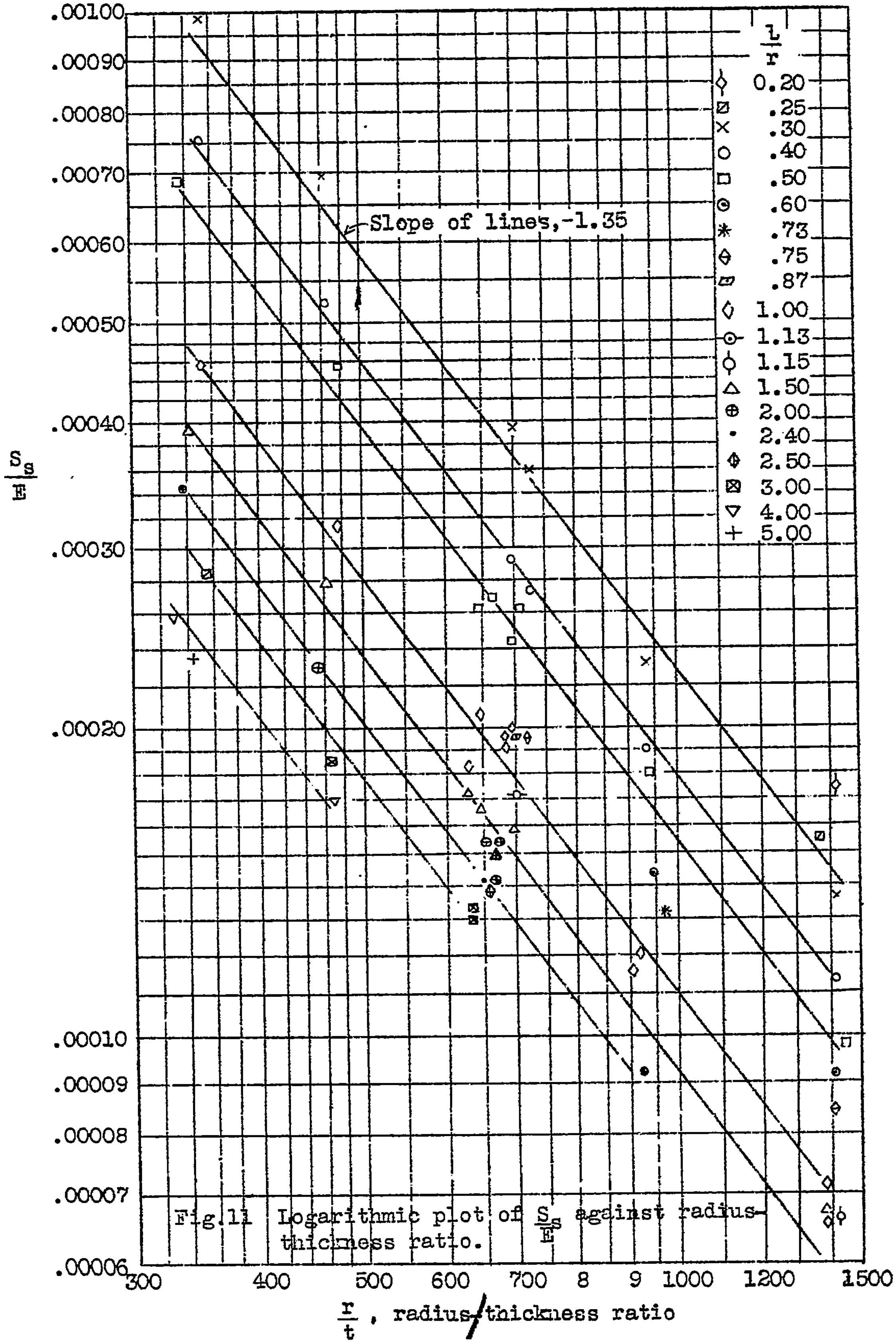
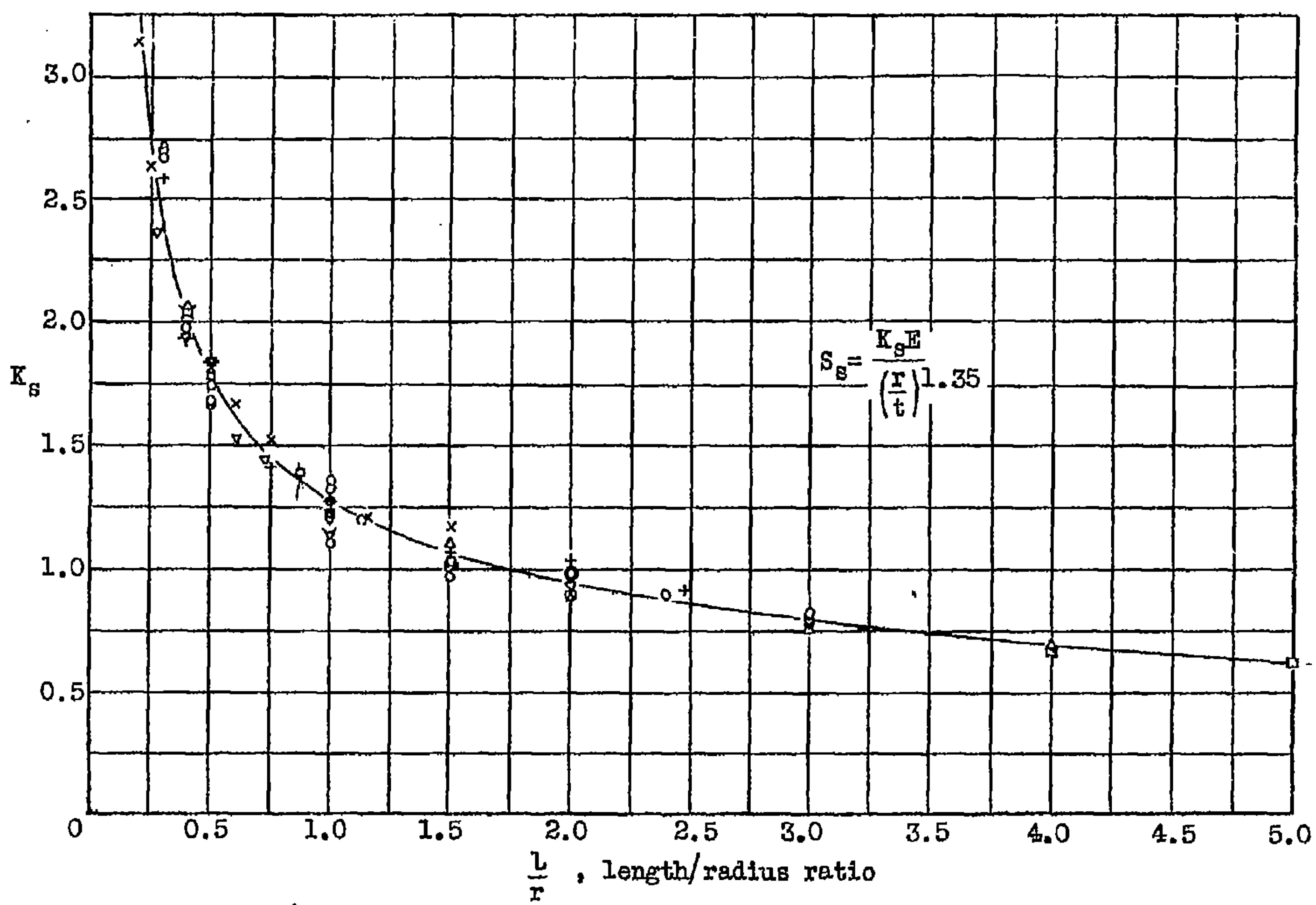


Fig.11 Logarithmic plot of $\frac{S_s}{E}$ against radius-thickness ratio.



	Radius	Radius/thickness
□	7.5	329 to 354
△	7.5	452 to 472
○	7.5	630 to 700
+	15.0	658 to 721
▽	15.0	903 to 963
×	15.0	1364 to 1428

Fig.12 Variation of K_s with length/radius ratio.